CHALMERS

Modelling, failure modes prediction and optimization of gear shifting mechanism

Application to heavy vehicle transmission systems

MUHAMMAD IRFAN

Thesis submitted for the degree of Doctor of Philosophy in Solid and Structural Mechanics at the Department of Mechanics and Maritime Sciences
Chalmers University of Technology, Gothenburg, Sweden

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ABSTRACT

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A generic synchronizer modelled with five degrees of freedom and comprising three rigid bodies is studied to understand the gear shifting process. To get insight into the complete gear shifting process, detailed kinematic description of the phases and sub-phases is given. Nature of bodies' interaction is studied. A mathematical model is developed based on Constrained Lagrangian Formalism. The developed model is validated against test rig data. After sensitivity analysis, optimization is performed based upon the developed model. Synchronization time and speed difference at end of the main phase of synchronization process are chosen as objective functions. Parameters are cone angle, cone coefficient of friction, applied shift force, blocker angle, blocker coefficient of friction, cone radius, gear moment of inertia and ring moment of inertia. Several cases of the synchronization process are studied under different scenarios of master/slave and different operating conditions. Further analysis of results obtained from Pareto optimization clarifies the degree of influence of the input parameters.

To identify the failure modes, the gear shifting mechanism is modelled on GT-Suite software. System response characteristics are chosen to observe the failure modes. At failure modes occurrence limits of values of design parameters are identified. With these limits genetic algorithm based routine is applied to the optimization. The synchronization time is selected as an objective function to be minimized. At first step seven parameters are considered as varying parameters for optimization. At second step seventeen design parameters are optimized for six cases at master/slave settings with conditions of nominal, road grade and driveline excitation. Because of the minor differences between the optimization results average values of the parameters are taken as optimal values for all cases. It is shown that the obtained optimized values of design parameters are robust with respect to different driving conditions.

Keywords: Synchronizer, gear shifting, constrained Lagrangian formalism, GT-Suite modelling, sensitivity analysis, optimization, failure modes prediction.

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Cover:
Figure illustrates the exemplary cases of the gear shifting mechanism.

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PREFACE

The research work has been carried out since 3rd of March 2014 at Division of Dynamic, at Department of Mechanics and Maritime Sciences, Chalmers University of Technology. The work is a part of the project of transmission cluster with collaboration of AB Volvo, Scania CV AB, Royal Institute of Technology, Chalmers University of Technology and VINNOVA. The project is funded by AB Volvo, Scania CV AB, Chalmers University of Technology and VINNOVA.

First of all I would like to thank my ALLAH ALMIGHTY WHO is the most GRACIOUS and the most MERCIFUL.

I would like to thank my supervisor Viktor Berbyuk who has led the research with his experience and to produce the impressive research. I am also thankful to my co-supervisor Håkan Johansson for his interest in the work and his efforts to increase quality of the work. Collectively I would like to express my sincere gratitude to my both supervisors for contribution of their ideas to increase quality of the research project.

A special thanks to Magnus Andersson from Volvo Cars, Ulf L Sellgren at Royal Institute of Technology, Daniel Häggström and Kenth Hellström at Scania CV AB for their fruitful discussion.

I take this opportunity to express my deepest gratitude to my mother, my father, other family members, and my teacher Al-Shah Mazhar Fareed Subhaani for their support and motivation to continue my higher education.

Gothenburg, January 2019
Muhammad Irfan

THESIS

This thesis includes an extended summary and the following appended papers:

Paper A	M. Irfan, V. Berbyuk and H. Johansson, (2015), "Modelling of heavy vehicle transmission synchronizer using constrained Lagrangian formalism", *In Proc. of the International Conference on Engineering Vibration,* Ljubljana, 7 - 10 September ; [editors Miha Boltežar, Janko Slavič, Marian Wiercigroch]. - EBook. - Ljubljana: Faculty for Mechanical Engineering, 2015 p. 28-37.
Paper B	Irfan, M., Berbyuk, V., Johansson, H., (2016), "Dynamics and Pareto Optimization of a Generic Synchronizer Mechanism", in *Rotating Machinery, Hybrid Test Methods, Vibro-Acoustic & Laser Vibrometry,* Proceedings of the 34th IMAC, A Conference and Exposition on Structural Dynamics 2016*, Editors James De Clerck and David S. Epp, Volume 8, pp. 417-425, 2016, Springer,* ISBN: 978-3-319-30084-9, http://dx.doi.org/10.1007/978-3-319-30084-9_38.
Paper C	M. Irfan, V. Berbyuk and H. Johansson, "Performance improvement of a transmission synchronizer via sensitivity analysis and parametric optimization," *Cogent Engineering,* vol. 05, nr 01, pp. 1-46, 14 Jun 2018, DOI: 10.1080/23311916.2018.1471768.
Paper D	M. Irfan, V. Berbyuk and H. Johansson, "Failure modes and optimal performance of a generic synchronizer," *The 5th Joint International Conference on Multibody System Dynamics*, Lisboa, Portugal, 24-28 June 2018.
Paper E	M. Irfan, V. Berbyuk and H. Johansson, "Minimizing synchronization time of a gear shifting mechanism by optimizing its structural design parameters," *submitted for international publication*.

The appended papers were prepared in collaboration with the co-authors. The author of this thesis was responsible for major progress of the work in preparing the papers, i.e. took part in planning the papers, developing the theory, performing all implementations and numerical calculations, analysis of the results and writing.

In addition to papers A-E the following report and conference paper have also been part of the research in this PhD project which are not included in this thesis:

Irfan, M., Berbyuk, V., Johansson, H., "*Constrained Lagrangian Formulation for modelling and analysis of transmission synchronizers*," Report 2015:05, Department of Applied Mechanics, Chalmers University of Technology, Gothenburg, 2015.
http://publications.lib.chalmers.se/publication/233233

M. Irfan, V. Berbyuk and H. Johansson, "Verification of a transmission synchronization model," Conference contribution, Svenska Mekanikdagarna, Uppsala, p. 48, 12-13 June 2017.

CONTENTS

Part I

Extended Summary

1 Introduction and motivation

Automotive industry is struggling to meet the future demands in order to reduce carbon dioxide emissions particularly in case of heavy vehicles. In this regard different components of the heavy vehicle (engine design, drivetrain etc.) are under consideration to update. Gearbox, including gear shifting mechanism, is a crucial part of the drivetrain. The gear shifting mechanism (synchronizer) needs to be quick, smooth and energy efficient to meet the development of new technology for the automotive industry. Before describing about the study in following paragraphs a literature review upon the transmission synchronizer is presented.

A numerical method is proposed for pre-synchronization and to increase performance of the synchronizing component in [1]. The numerical method helps to avoid difficult analytical methods and to minimize the traditionally exhaustive physical testing. A multi-physics FEM software, a multi-physics FEM software, is used. Presence of the grooves on surface of the synchronizer proves to be more important than design of the grooves. Failure of the molybdenum coated synchronizer is identified by applying different load conditions in [2]. Four parameters are used to identify the failure. These parameters are synchronized kinetic energy, synchronization power, focal surface temperature and average surface temperature. A threshold is also predicted which inform about longer or shorter life of the synchronizer. A measurement method of average surface temperature is also presented. In [3] a generalized thermomechanical model is developed for analysis and design of the synchronizer. The finite element tool Abaqus/Standard is used. Effects of external loads and design parameters on temperature transient in friction lining are studied. The model is verified with analytical means and validated with measurements data. The model is applied on different design cases. A friction model of lubricated molybdenum-steel contact is developed in [4]. An effort is made for the complex transient synchronization process where the parameters have strong mutual dependences. The model is based upon measurements of test rig and FEM simulations. Another simplified model is developed to quickly estimate friction coefficient and temperature.

A new design of gear shifting mechanism for manual transmission electric vehicle is proposed to increase the gear shift efficiency in electric cars [5]. A combination of synchromesh and zero shift mechanism is introduced. Function of manual transmission clutch is expected to replace with motor torque setting. The new design is simple for manufacturing, easy for installation and easy to use by the drivers. In [6] a multibody dynamic model of the synchronizer is presented. Contact forces between engaging teeth and frictional torque are measured by experiments. Lubricated friction model is established to simulate the friction forces. Shift force is varied by a pneumatic model. The model is validated with experimental data. A multibody dynamic model of the synchronizer is proposed to predict the synchronization time [7]. The model is validated with single cone and two cones experimental setup at different angular speeds, inertia, friction coefficient and loading conditions. The model is also verified with analytical results.

In Lovas [8] studied the Borg-Warner synchronizer. Design, working and problems of the synchronizer are presented. A mathematical model is developed based upon different models of sub phases of the synchronization with boundary conditions. An equivalent cone synchronizer is simulated instead of multi-cones synchronizer. The model is validated with test bench measurements. Effects of different factors are studied on double bump. Stick-slip phenomenon is studied at grooves level of the conical surfaces. Hypothesis of the synchronizer ring is proved by measurements and simulation results. Stiffness and damping of the system are determined. Abel et al. in [9] presented typical simulation and optimization challenges for the comfort of automatic shifting. The model is validated with experimental measurements. Unknown parameters in the model are adjusted with typical ranges of parameters. The model can provide access to any physical quantity. The model suggests early design phases before a prototype. In short the model speeds up the design process of the synchronizer. In [10] higher frictional torque of the synchronizer is presented without changing the shift force. The concept of double indexing is presented which can reduce the synchronization time. Mathematical, analytical, simulated and test rig results are presented.

In [11] a multibody dynamic model is developed. The model describes dynamic analysis at different operational conditions. Single, double and triple cones synchronizers are proposed to validate with the test rig measurements. An analytical model estimates the synchronization time. A sensitivity analysis based upon the effect of tolerance dimension on the dynamic behavior is also studied. A physics-based component level model provides explanation of contact and impact dynamics in [12]. The model is integrated into a powertrain system model after verification with AMESim block. The analysis defines sliding speed during gear shifting in a 6-speed automatic transmission. The model is validated with dynamometer data. In [13] Modelica®-based hydro-mechanical model is presented to optimize the shift quality. The shifting behavior is validated with experimental data. The model presents a detail about gear shift process of automated manual transmission, hydro-mechanical actuators and clutch disk spring displacement-force characteristics. The gear shifting process is modelled in Matlab/Simulink in [14]. The model is verified with an Adams model and validated with experimental data. Parameters of the gear shifting mechanism are optimized based upon the quality measures which are the shift force and control accuracy of the driving motor speed. The models are presented to study the sensitivity of parameters of the gear shifting mechanism in [15]. The parameterized models are developed in MSC Adams. The parameters are analyzed against shift time and peak force.

In addition to above mentioned papers gear shifting mechanism remains a research focus area [22-35].

1.1 Research focus and questions

Although in the above studies a numerical method, a friction model of lubricated molybdenum-steel contact, a thermomechanical model, failure of the molybdenum coated synchronizer, a new design of gear shifting mechanism for manual transmission electric vehicle, a multibody dynamic model of the synchronizer, design, working and problems of the Borg-Warner synchronizer, typical simulation and optimization challenges for the comfort of automatic shifting are presented, still such a model is missing which can provide a ground to study the gear shifting process in detail, performance at different driving conditions, failure modes, degree of influence of the structural design parameters and optimization. In this concern the mathematical model based on constrained Lagrangian formalism and the model in GT-Suite software are developed. Failure modes are predicted and sensitivity analysis is performed. The gear shifting process is studied with four phases and eleven sub-phases. Optimized values of the structural design parameters are obtained at different driving conditions: nominal, excitation and road grade.

In particular the following research questions are in focus:

Is it possible to understand complex movement of the bodies in the synchronizer mechanism through sub-phases of the developed models?

What is the degree of influence of structural design parameters of the synchronizer upon the gear shifting process?

How can failure modes be predicted by selecting suitable values of the structural design parameters?

Which parameter values can be considered optimal for the performance at different driving scenarios?

1.2 Thesis outline

In chapter 1 a brief summary of the work, references of the research papers on synchronizer, questions about the conducted research are outlined. In chapter 2 and 3 explanation of generic synchronizer, synchronization process are given and later on in chapter 4 the applied methodology to study the synchronization process is explained and the synchronizer is modelled in GT-Suite software. Some results of developed model of synchronization are presented in chapter 5. Sensitivity analysis is presented in chapter 6. A description about failure modes prediction is given in chapter 7. In chapter 8 problem statement of the Pareto optimization is given along with some results obtained. In chapter 9 appended papers are summarized briefly. The last chapter discussed conclusions and outlook of the future work. Afterwards the papers A-E are attached.

2 Transmission synchronizer

The chapter explains about a gear shifting mechanism, number of rigid bodies and structural design of each body.

Transmission system transmits torque from engine to wheels of a vehicle as shown in Figure 1. The gearbox with different gear numbers of the transmission system matches the torque produced by engine with vehicle desire speed. It is clear that shifting between the gears is required for smooth driving of the vehicle. The mechanism used for gear shifting is called synchronizer. Still there is demand of automotive industry to complete the gear shifting process as quick as possible with smoothness. Before proceeding to analyze the gear shifting process for quickness and smoothness, in paper A the process is explained in detail together with the mathematical model. A short description of the gear shifting mechanism and the gear shifting process is also given below.

The synchronizer can have different number of bodies with different shapes, and a variety of different concepts are used in vehicles today. To avoid specific existing commercially available synchronizers, this thesis will concern a generic synchronizer as shown in Figure 2. A generic synchronizer is a simplified multibody system of a gear shifting mechanism. It consists of three bodies which are called engaging sleeve, synchronizing ring and gearwheel. In this multibody system the sleeve and the gearwheel need to connect with each other through meshing of their teeth to shift a gear. The sleeve and the gearwheel cannot be connected directly because both bodies are rotating at different rotational speed. Here the ring plays its role to bring both bodies at the approximately same rotational speed. Then teeth of the bodies mesh with each other and complete the gear shifting process.

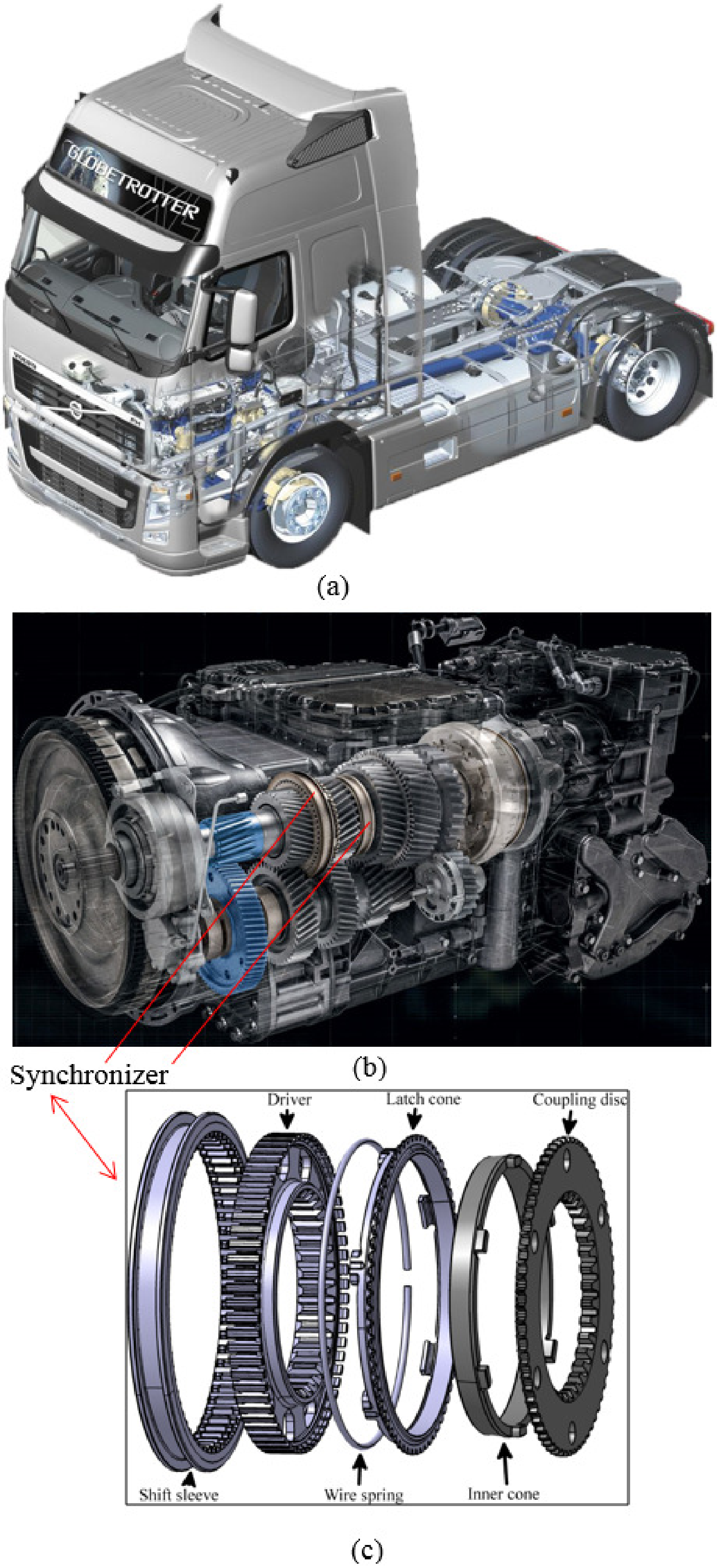

Figure 1: Description and position of the synchronizer in a vehicle [16, 17, 18].

The gear shifting mechanism can have different design structure to perform the gear shifting process. In this generic synchronizer the sleeve and the ring have frictional cones, the ring and the gear have blocking teeth, and the gear and the sleeve have engaging teeth.

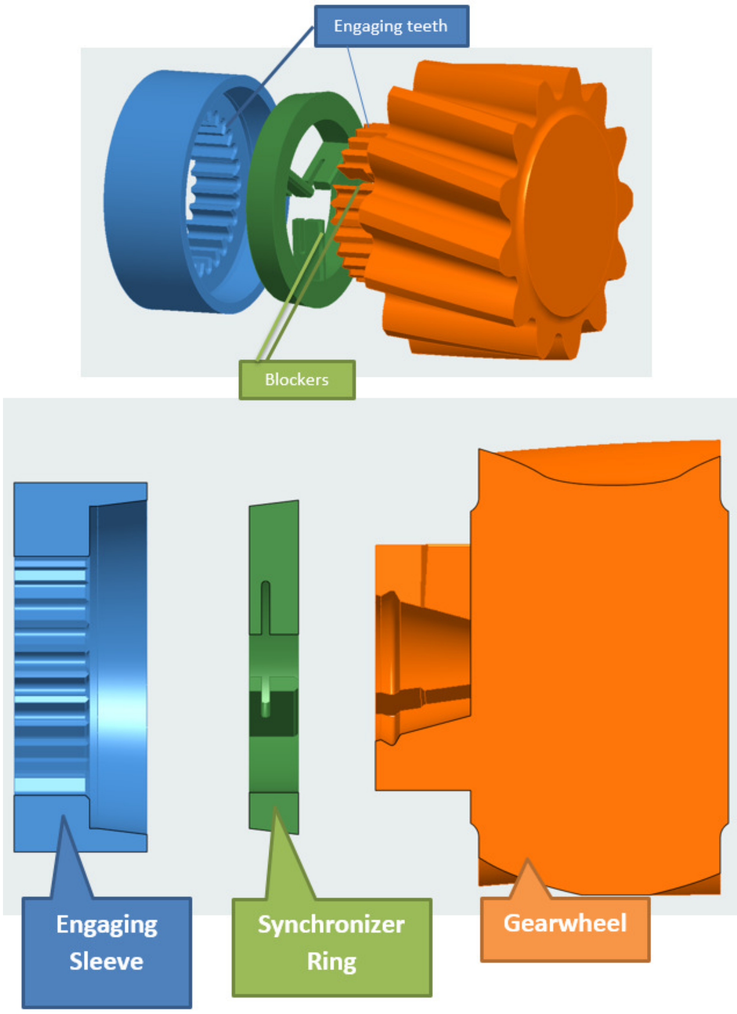

Figure 2: 3D and 2D front view of the generic synchronizer.

Design of the gear shifting mechanism is explained which has considered for the study. Detail description of the mechanism can be found in paper A and [19]. Working of the transmission synchronizer is explained in the section 3 (transmission synchronization).

3 Transmission synchronization

Performance of the gear shifting mechanism is described in this chapter.

The gear shifting process, synchronization, is a complex process because of the structural design and interaction of the bodies. To understand easily the synchronization process is analyzed in four phases which are further divided into several sub-phases (paper A) as shown in Figure 3. At beginning of the gear shifting process shift force is applied on the sleeve and it starts to move axially. The ring rotates relative to the sleeve because of the oil frictional torque between the cones. The ring gets its indexing position during sub-phase 1a to block the sleeve. The sleeve moves freely during sub-phase 1b and comes to the position from where sufficient resistance starts to push the ring axially. The sleeve and the ring move axially together during sub-phase 1c. The ring comes in contact with the gearwheel through the blocking teeth. Now the oil squeeze out between the cones during sub-phases 1d and 1e. The sleeve does not move axially during sub-phase 1f and phase 2. The ring and gearwheel are rotating with same rotational speed and the sleeve is rotating with different rotational speed. The cones are sliding over each other with friction. Sliding frictional torque reduces speed difference between the sleeve and the gearwheel. Two opposite and axial net forces are main agents at this moment to block the sleeve and to reduce the speed difference. One is at blocking teeth and second is at cones contact. In start of the contact, the axial force at blocking teeth contact is higher than the axial force at cones contact. When the axial force at cones contact becomes greater than the axial force at blocking teeth, the sleeve starts to move axially again and the speed difference approaches zero. During sub-phase 3a the sleeve moves freely and its teeth approaches engaging teeth of the gear. The sleeves engaging teeth turns out the gear to move axially during sub-phase 3b. At end of the process during sub-phase 3c and phase 4 engaging teeth of the sleeve and the gear come in contact and the gear shifting process is completed. More detailed description of kinematics of synchronization process is presented in Paper A and in [19].

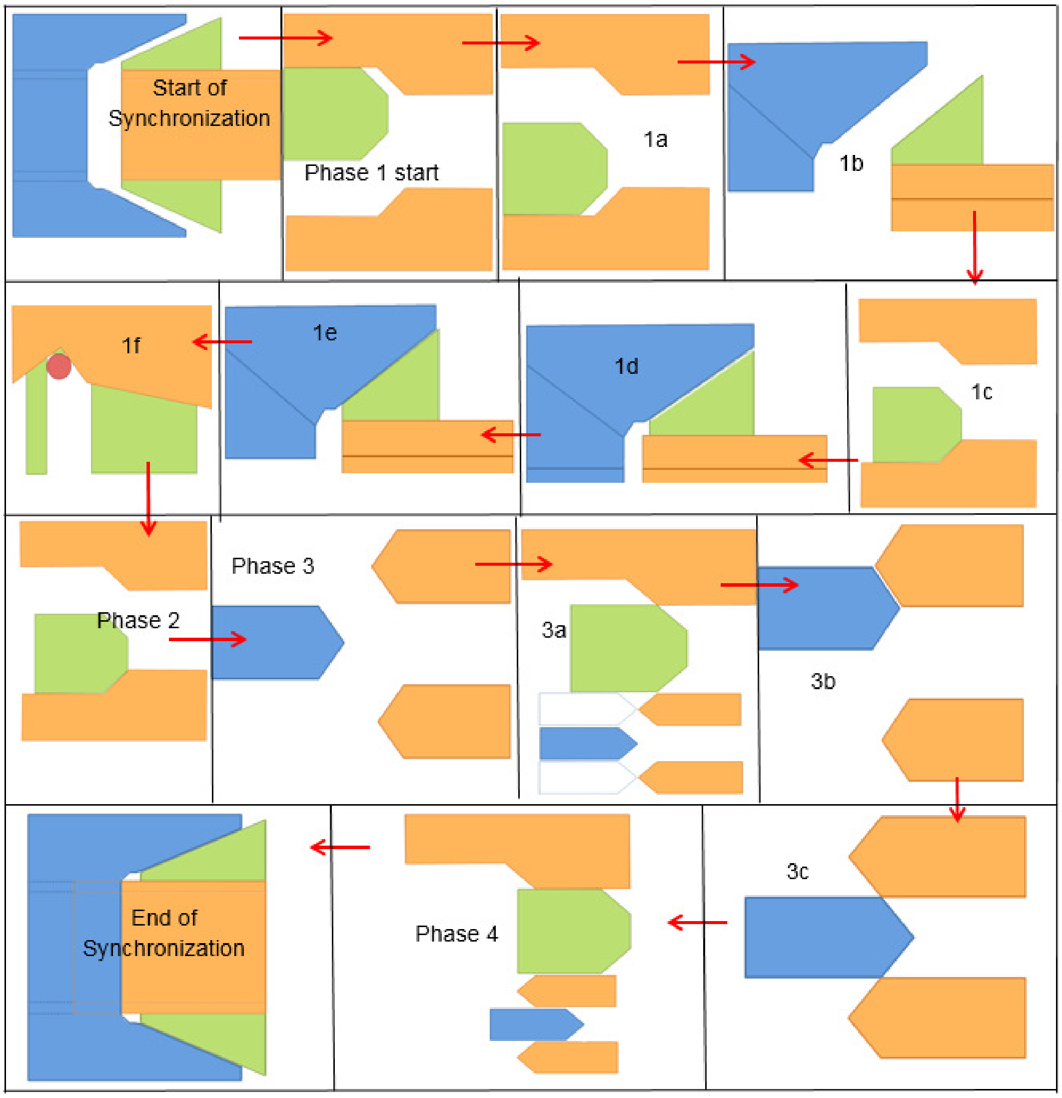

Figure 3: Sub-phases of the generic synchronization (paper A).

4 Methodology

A constrained Lagrangian Formalism (CLF) is applied upon the gear shifting process to develop a mathematical model. GT-Suite software is used to develop another model of the synchronizer.

4.1 Constrained Lagrangian Formalism

The methodology used to develop the mathematical model for synchronization process of the generic synchronizer is given in paper A and [19]. Here we present elements of constrained Lagrangian formalism used to derive the equations of motion of a gear shifting mechanism. As it is described in introduction of chapter 1, still there is a need of such kind of a model which can describe the gear shifting process thoroughly. The model could also provide opportunity to update the mechanism for the new design. The model could connect with other relevant models of the transmission system of vehicle as well. So in this regard a constrained Lagrangian formalism is proposed which explain in detail the gear shifting process in phases and sub-phases.

Let's suppose multibody system comprises n bodies as shown in Figure 4. Some of the bodies are connected through kinematic constraints. Motion of some bodies get influence by other bodies.

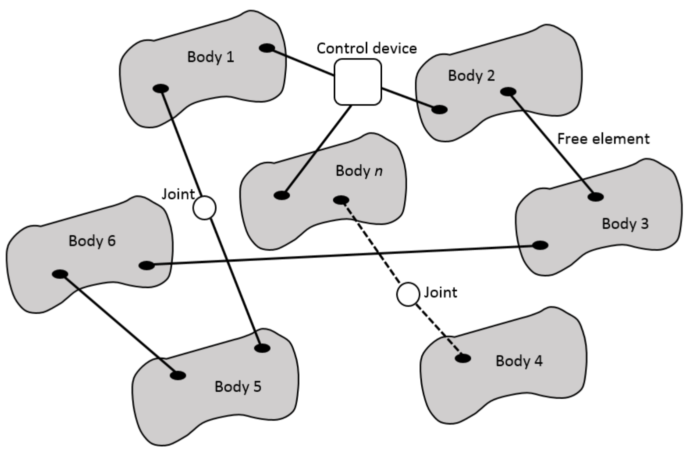

Figure 4: Multibody system.

Let the generalized coordinates of the multibody system is represented by vector $\boldsymbol{q} = [q_1 \;\; q_2 \;\; q_3 \; . \; . \; . \; q_n]^T$ where n is number of generalized coordinates. Let also assume that the set of independent constraints is imposed on the system that can be represented by the following equations

$$\boldsymbol{C} = \left[C_1(\boldsymbol{q},t) \;\; C_2(\boldsymbol{q},t) \;\; . \; . \; . \; C_{n_c}(\boldsymbol{q},t)\right]^T = \boldsymbol{0}\,. \tag{1}$$

If equations of the constraints can be written in the above vector form, i.e. $\boldsymbol{C}\,(\boldsymbol{q},t) = \boldsymbol{0}$, the constraints as well as the respective system are called holonomic. In the holonomic system if t appears explicitly, the system is said to be rheonomic whereas if t does not appear explicitly, the system is said to be scleronomic. The constraints which can't be expressed in the form (1) and can be written as $\boldsymbol{C}\,(\boldsymbol{q},\dot{\boldsymbol{q}},t) = \boldsymbol{0}$ are called nonholonomic constraints.

According to the orthogonality theorem and Lagrange multiplier theorem [20, 21] the constraint force can be written as

$$\boldsymbol{F}^c = -\,\boldsymbol{C}_{\boldsymbol{q}}^T \;\boldsymbol{\lambda}\;. \tag{2}$$

Here $\boldsymbol{\lambda}$ is vector of the Lagrangian multipliers and $\boldsymbol{C}_{\boldsymbol{q}}^T$ is the constraints Jacobian of the system.

By introducing the Lagrangian L as

$$L = T - V\,,$$

where T is the kinetic energy, V is the potential energy including strain energy and potential of any conservative external forces, the equations of motion of the multibody system can be written as follows

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{\boldsymbol{q}}}\right)^T - \left(\frac{\partial L}{\partial \boldsymbol{q}}\right)^T + \boldsymbol{C}_{\boldsymbol{q}}^T\,\lambda = \boldsymbol{Q}\,, \tag{3}$$

$$\frac{\partial L}{\partial \dot{\boldsymbol{q}}} = \begin{bmatrix} \frac{\partial L}{\partial \dot{q}_1} & \frac{\partial L}{\partial \dot{q}_2} & \cdots & \frac{\partial L}{\partial \dot{q}_n} \end{bmatrix},$$

$$\frac{\partial L}{\partial \boldsymbol{q}} = \begin{bmatrix} \frac{\partial L}{\partial q_1} & \frac{\partial L}{\partial q_2} & \cdots & \frac{\partial L}{\partial q_n} \end{bmatrix},$$

$$\boldsymbol{C}_{\boldsymbol{q}} = \begin{bmatrix} C_{11} & C_{12} & \cdots & C_{1n} \\ C_{21} & C_{22} & \cdots & C_{2n} \\ \vdots & \vdots & \ddots & \vdots \\ C_{n_c1} & C_{n_c2} & \cdots & C_{n_cn} \end{bmatrix},$$

$$\boldsymbol{Q}^{\boldsymbol{T}} = [Q_1 \quad Q_2 \quad \dots \quad Q_n]\,,$$

where $\boldsymbol{Q}$ is a vector of the system generalized forces.

The equation (3) together with constraints (1) describe the motion of constrained multibody system and constitutes constrained Lagrangian formalism used to model the synchronization processes.

4.2 Modelling of synchronizer in GT-Suite software

A second model of the gear shifting mechanism is developed in GT-Suite v2016 to verify the CLF based model, to be used in the failure modes prediction and to get opportunity to optimize structural design parameters efficiently.

In Figure 5 model of the gear shifting mechanism in GT-Suite has three rigid bodies; sleeve, ring and gear. The sleeve and the ring can rotate about one axis and translate along the same axis. The gear can only rotate about the same axis. The sleeve has blocking teeth and engaging teeth. The ring has blocking teeth and the frictional cone. The gear has frictional cone and engaging teeth. At start of the gear shifting process the sleeves translates and make a contact with the ring through blocking teeth. Then the sleeve and the ring start to rotate and translate together. After a while the ring cone comes in contact with the gear cone. The sleeve and the ring do not translate further and the frictional cones slide over each other. Speed difference between the sleeve and the gear reduces because of frictional torque of the sliding cones. When the speed difference approaches zero, the sleeve translates again and it’s engaging teeth leaves the ring behind. At end of the process engaging teeth of the sleeve and the gear come in full contact.

In section 7 some of the results of failure modes from the model are presented (see also paper D).

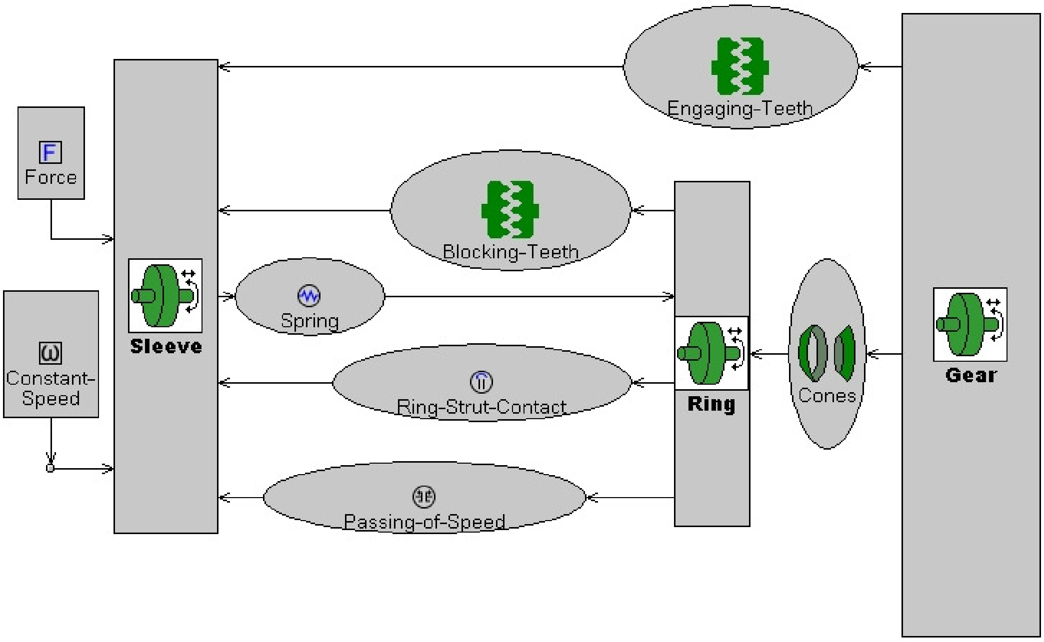

Figure 5: Modelling of gear shifting mechanism in GT-Suite (paper D).

5 Performance diagram

The mathematical model is developed and the gear shifting mechanism is described. Working of the mathematical model is evaluated in this chapter. For understanding of the gear shifting process performance diagram at nominal condition with sleeve as a master (constant rotational speed) is presented.

The developed mathematical model of a generic synchronizer mechanism is validated in paper A and used for analysis of dynamics of synchronization processes for different scenarios. The gear shifting process is simulated by the mathematical model where the sleeve is considered as a master and gear is considered as a slave at nominal condition. Values of the parameters to simulate the gear shifting process are shown in Table 1. In Figure 6 shift force is applied on the sleeve with continues rate of force till end of phase 2 from point *a* to point *h*. During phase 3 and phase 4 the shift force approaches to zero from point *h* to point *i*. The sleeve has constant rotational speed throughout the gear shifting process from point *f* to point *g*. The ring and the gear have same rotational speed as shown from point *j* to point *k*. The sleeve moves axially during phase 1 from point *a* to point *b* but does not move axially during phase 2 till point *c*. The sleeve axial speed during phase 3 and phase 4 is higher than the phase 1 because the sleeve covers larger displacement during phase 3 within shorter time than the covered displacement during phase 1. The gear rotational speed increases during phase 1 and phase 2. At end of the phase 2, speed difference becomes zero. During phase 3 the sleeve turns the gear to moves axially again. This movement of the gear is shown from point *m* to point *n*. During phase 4 the gear and the sleeve rotate with same rotational speed. More results of simulation of dynamics of a generic synchronizer for different scenarios and conditions (transmission vibrations, road grade, others) are presented in details in Papers A-E and [19].

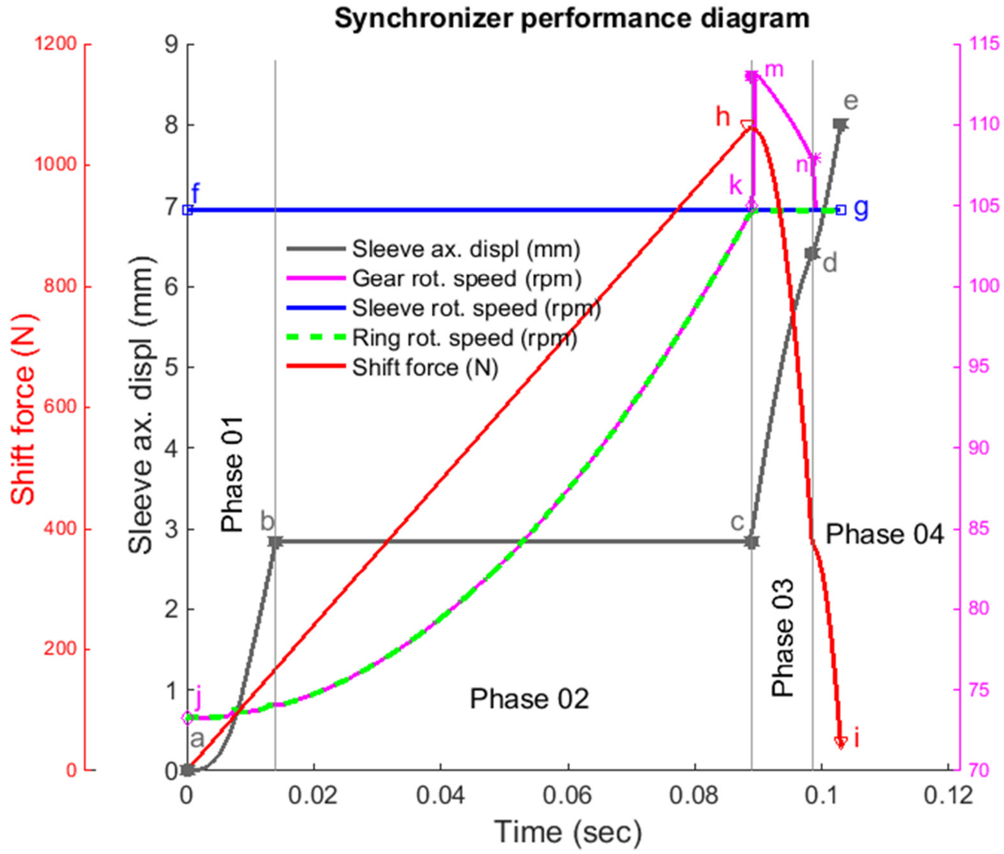

Figure 6: The generic synchronizer performance diagram (paper C).

Next question after studying performance of the synchronizer by the developed mathematical model is that how the performance will vary by changing parameters of the synchronizer? By the sensitivity analysis, effects of the parameters are studied in chapter 6.

Variable	Name	Variable	Name
$m_s = 1.5$ kg	Sleeve mass	$m_r = 0.5$ kg	Ring mass
$I_g = 0.2$ kgm^2	Gear moment of inertia	$I_s = 0.01$ kgm^2	Sleeve moment of inertia
$I_r = 0.004$ kgm^2	Ring moment of inertia	$r_\beta = 0.07$ m	Blocking teeth radius
$r_\alpha = 0.1$ m	Cones mean radius	$d_\alpha = 2$ mm	Cones Initial clearance
$b = 5$ mm	Cones contact length	$B_{ang} = 5^o$	Angle clearance
$\beta = 60^o$	Blocking teeth angle	$\mu_\alpha = 0.17$	Cones friction coefficient
$\alpha = 7^o$	Cone angle	$\mu_\beta = 0.09$	Blocker friction coefficient
$C_{r\alpha} = 0.002$	Ring sliding friction	$C_{s\alpha} = 0.02$	Sleeve sliding friction
$\omega_{s_o} = 1000$ rpm	Sleeve rotational speed	$\omega_{g_o} = 700$ rpm	Initial gear rotational speed
$F_{shf} = 1000$ N - Shift force			

Table 1: Values of parameters of the generic synchronizer used for simulation (paper D).

6 Sensitivity analysis

The chapter presents sensitivity analysis where variation of the synchronization time is studied by varying the design parameters. The analysis shows degree of influence of parameters upon the synchronization time and complexity of the dependency of eight parameters.

One of the main objectives under all circumstances is to shift the gear as quick as possible. But it is also demanded to make the gear shift as smooth as possible. A desirable objective of the generic synchronizer is a minimum synchronization time. Structural design parameters and shift force are varied to study their effects upon the synchronization time. Reasonable bounds of the parameters are selected as given in paper D.

The developed mathematical model has been used for sensitivity analysis of dynamics of synchronization processes. Sensitivity analysis with eight parameters is given in paper C where degrees of influence of eight parameters: cone angle, cone coefficient of friction, rate of shift force, blocker angle, blocker coefficient of friction, cone radius, gear moment of inertia and ring moment of inertia are analyzed with respect to synchronization time.

Figure 7 shows variation of the structural and control design parameters with respect to the synchronization time. The synchronization time increases with increasing cone angle, blocker angle, gear moment of inertia and ring moment of inertia but the synchronization time decreases with increasing cone friction, rate of shift force, blocker friction and cone radius. In this sensitivity analysis while varying a particular parameter to obtain variation of the synchronization time, values of rest of the parameters are remained constant. The Figure 7 shows the gear shifting process is not a simple process with respect to variation of the parameters. Figure 7 shows few parameters have less degree of influence than others. For example cone radius and ring moment of inertia have less degree of influence upon the synchronization time. More detail about sensitivity analysis can be found in paper B and C.

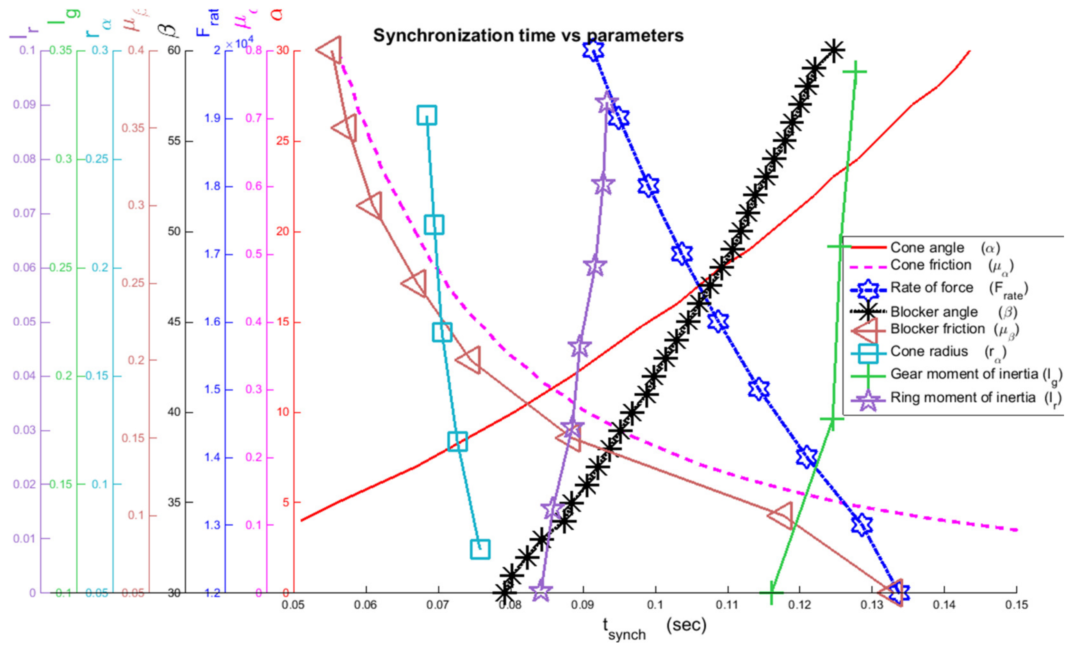

Figure 7: Effect of eight parameters on synchronization time (paper B).

7 Failure modes prediction

In this chapter failure modes are predicted by performing the sensitivity analysis in GT-Suite software.

Purpose of the generic synchronizer is to shift the gear. In case of failure the generic synchronizer does not shift the gear. The simplified generic synchronizer starts working by applying a reasonable shift force. In this study values of structural design parameters and shift force are varied to find out that at which values the synchronizer fails to perform. Three different types of failure modes are predicted: clashing, blocking and longer synchronization time. In failure mode of clashing the engaging teeth come in contact and bounce back in a short time as shown in Figure 8 (a). In blocking failure mode the teeth block each other at contact and do not move further as shown in Figure 8 (b). When the synchronizer takes longer time to shift the gear which is not acceptable time, this performance mode is also called failure mode as shown in Figure 8 (c)-(d). Sleeve axial displacement, ring axial displacement, speed difference and relative rotation of the blocking teeth are monitored to identify the failure modes. Values of the parameters at failure modes are taken as limits of values for optimization. Further explanation about failure modes is given in paper D.

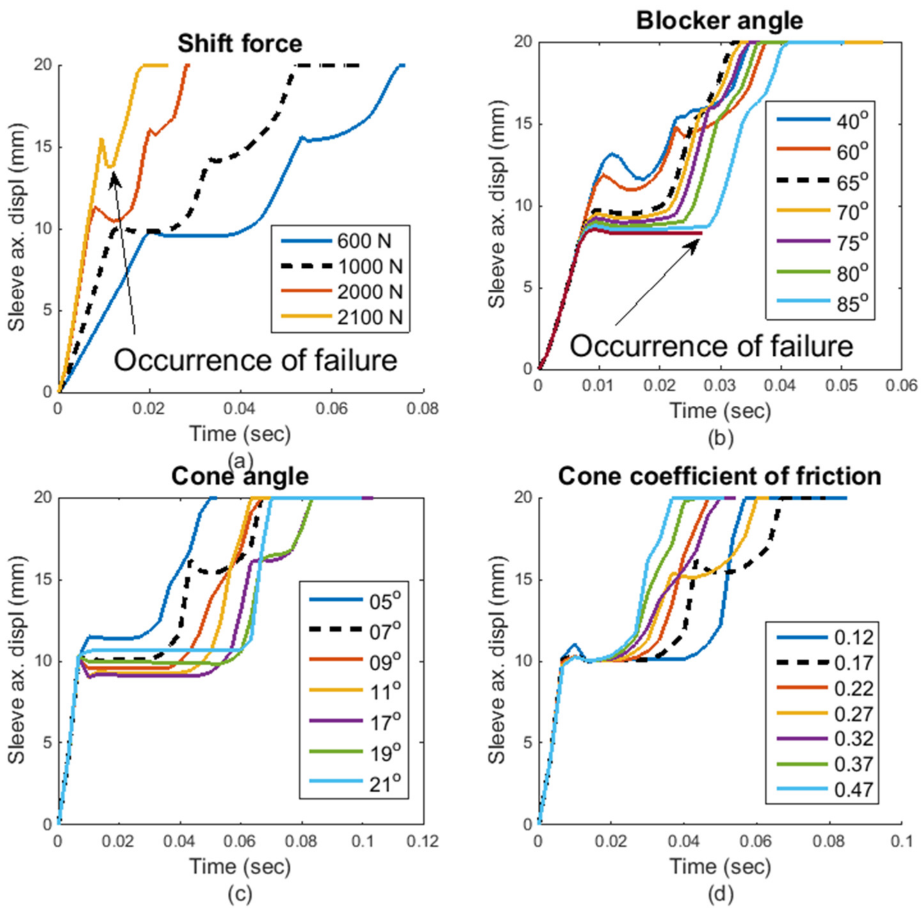

Figure 8: Failure modes prediction by monitoring the sleeve axial displacement (paper D).

8 Optimization of a generic synchronizer

Problem statement

The synchronization time, t_{synch}, is optimized by variation of the parameters. To obtain optimized performance of the synchronizer in terms of quick gear shifting it is necessary to determine the value of $\boldsymbol{X}$ which gives minimum of the function $t_{synch}(\boldsymbol{X})$ subject to GT-Suite model together with constraints. The mathematical statement of the optimization problem is formulated as follows.

$$\begin{cases} \min\limits_{\boldsymbol{X}} \left(t_{synch}(\boldsymbol{X})\right) \\ \boldsymbol{X}_l \leq \boldsymbol{X} \leq \boldsymbol{X}_u \end{cases},$$

where $\boldsymbol{X} = [\, F_s, \gamma,\, \beta_r, \beta_l, \alpha, \mu_\alpha, r_{min}, r_{max}]^T$. Here F_s is shift force, γ is indexing angle, β_r is blocking teeth right angle, β_l is blocking teeth left angle, α is cone angle, μ_α is cone coefficient of friction, r_{min} is cone minimum radius and r_{max} is cone maximum radius.

The generic synchronizer performs optimally at optimized values of the parameters. The genetic algorithm based optimization routine of GT-Suite software is applied on the transmission synchronization.

The synchronization time is plotted against the number of iterations as shown in Figure 9. The synchronization time decreases from 0.05 sec to 0.01 sec. Although the smallest gear shift time is reached but optimization continues to run to get clear crowded areas.

The synchronization time and the shift force and structural design parameters are plotted in Figure 10-11. Values of the control design and structural design parameters around more crowded areas are selected as optimized values because the GT-Suite model routine finds only these crowded areas again and again for the minimum synchronization time during searching of the optimal area from the design space. Approximate values of the control design and structural design parameters around the crowded areas, their percentage change from initial value and lower\upper bounds of the values are given in Table 2.

In Figure 10 the shift force, the indexing angle and the blocking teeth right and left angle are plotted. In Figure 10 (a) the crowded area is around 2400 N to 2500 N and from 0.01 sec to 0.02 sec. In case of the indexing angle there are almost three crowded areas in Figure 10 (b). Because lowest indexing angle is predicted as optimal angle during prediction of the failure mode therefore indexing angle at most left crowded area is selected as an optimal value. There are three crowded areas for synchronization time and three of them have no any sufficient difference. The lowest indexing angle of 4 degrees is selected as an optimal value because lower indexing angle supports minimum synchronization time. Practically blocker teeth are dealt with two angles; one is right angle and second is left angle. Before in this study for simplicity both angles are considered as same and studied with single value. But for optimization because of unavailability of such kind of simplicity in GT-Suite model the blocking teeth are studied with its two angles. In Figure 10 (c) and (d) based on the crowded areas at the minimum synchronization time, 50 degrees for right angle and 50 degrees for left angle of the blocking teeth are selected as optimal values. With similar kind of observations from Figure 11 (a-d) optimized values of the cone angle, the cone coefficient of friction, the cone maximum radius, the cone minimum radius and the spring force are given in Table 2. Initial values, average optimized values, percentage changes and variables bounds for values of the parameters and the synchronization time are given in Table 2.

From results of the optimization it can be concluded that by choosing optimized values of the parameters, the synchronization time can be minimized. Further seventeen parameters are considered to minimize the synchronization time through optimization. Detail can be found in Paper E.

<table>
<tr><th colspan="2">Objective function</th><th>No.</th><th>Independent variables</th><th>Initial values</th><th>Variables bounds</th><th>Average optimized values</th><th>Percentage change</th></tr>
<tr><td colspan="2" rowspan="2">Synchronization time (t_{synch}) (sec)</td><td>1</td><td>Shift force (N)</td><td>1000</td><td>300-2500</td><td>2500</td><td>150</td></tr>
<tr><td>2</td><td>Indexing angle (°)</td><td>4.6</td><td>2-10</td><td>4</td><td>13.0</td></tr>
<tr><td>Initial value</td><td>Optimized value</td><td>3</td><td>Blocking teeth right angle (°)</td><td>57.7</td><td>15-85</td><td>45</td><td>22</td></tr>
<tr><td>0.118</td><td>0.009</td><td>4</td><td>Blocking teeth left Angle (°)</td><td>50</td><td>15-85</td><td>35</td><td>30</td></tr>
<tr><td colspan="2">Percentage change</td><td>5</td><td>Cone angle (°)</td><td>7</td><td>5-15</td><td>5</td><td>28.6</td></tr>
<tr><td colspan="2" rowspan="3">92</td><td>6</td><td>Cone coefficient of friction</td><td>0.17</td><td>0.07-0.3</td><td>0.27</td><td>58.8</td></tr>
<tr><td>7</td><td>Cone max. radius (mm)</td><td>100.75</td><td>90-110</td><td>110</td><td>09</td></tr>
<tr><td>8</td><td>Cone min. radius (mm)</td><td>99.25</td><td>80-100</td><td>83</td><td>16.4</td></tr>
</table>

Table 2: Percentage change of input parameters after optimization (paper D).

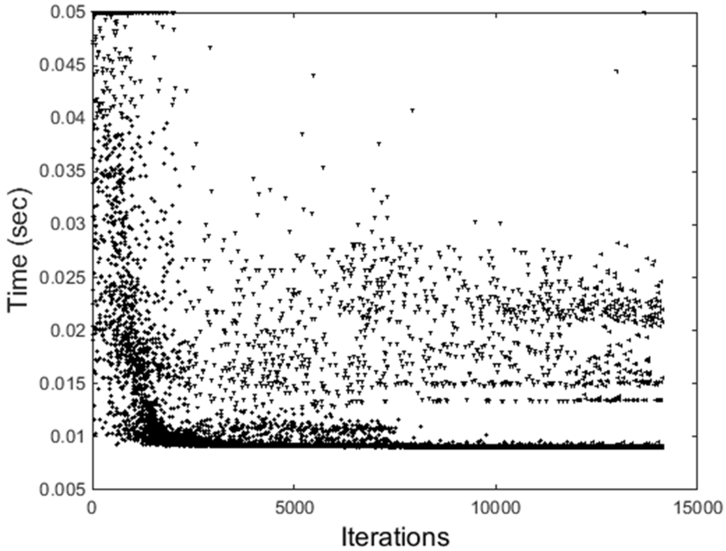

Figure 9: Behavior of synchronization time with number of iterations.

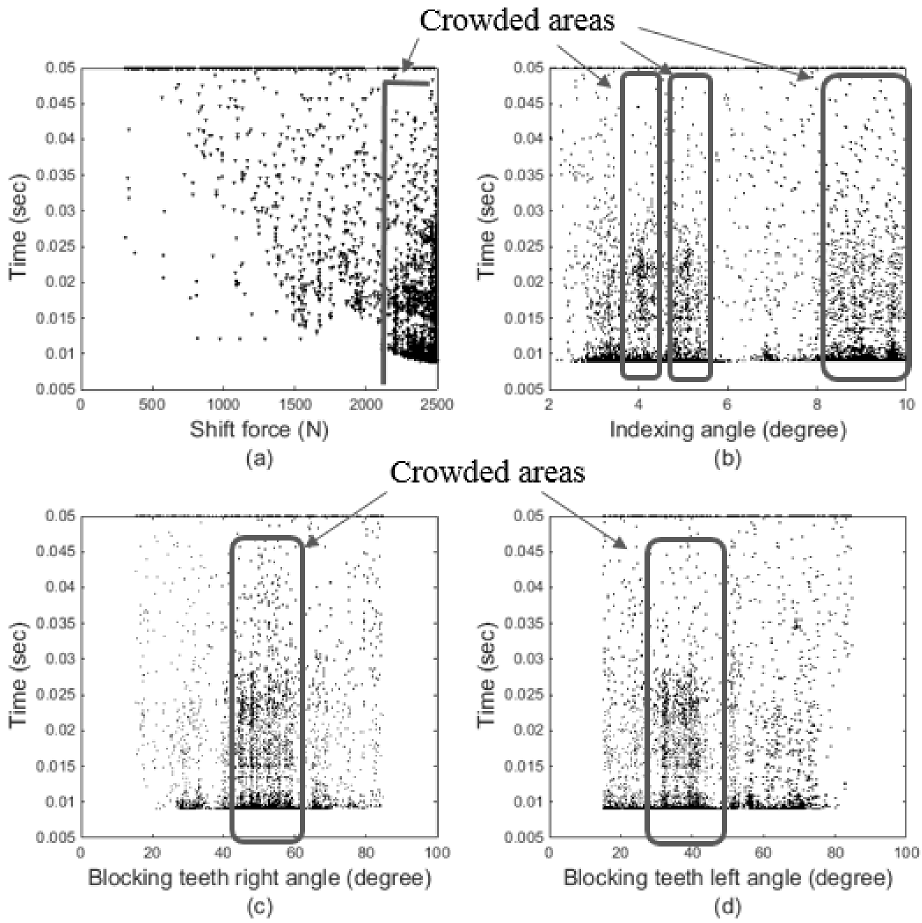

Figure 10: Correlation of optimized synchronization time with shift force and structural design parameters (paper D).

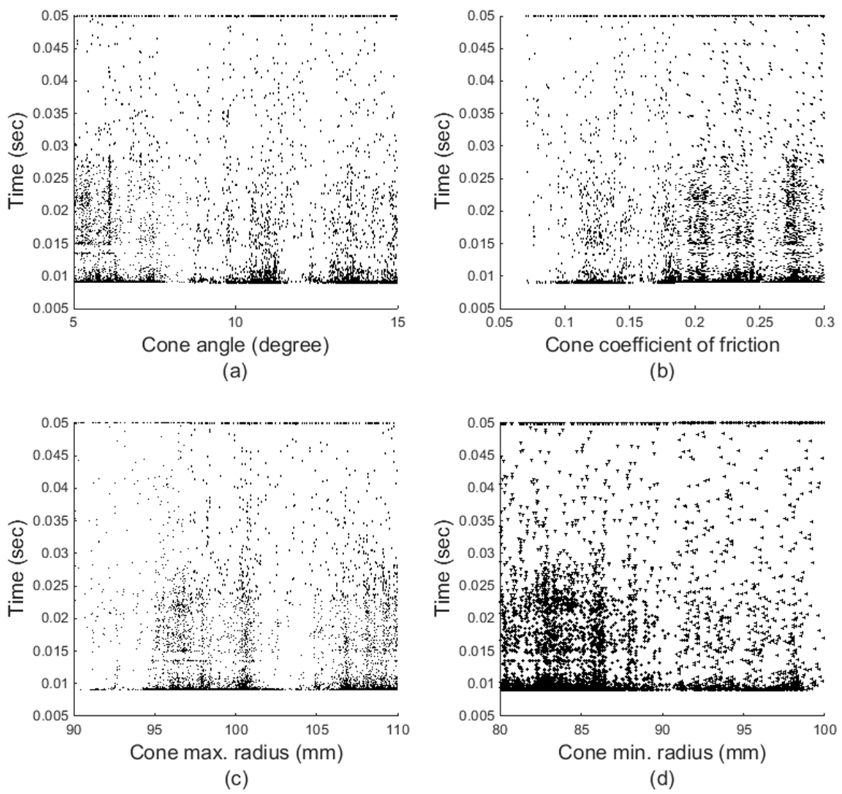

Figure 11: Correlation of optimized synchronization time with the structural design parameters (paper D).

9 Summary of Appended Papers

Paper A: Modelling of Heavy Vehicle Transmission Synchronizer using Constrained Lagrangian Formalism

The generic synchronization process is a complex phenomenon because of design of the bodies (sleeve, ring, gear) and their movement. Constrained Lagrangian formalism (CLF) explained the whole synchronization process in a unified manner with unilateral and bilateral constraints. There are different phases with sub-phases during gear shifting process and the CLF covers all the phases and sub-phases. Mathematical model of used CLF turns out into differential-algebraic equations to model the kinematics and kinetics of the generic synchronizer. The model is flexible to implement other relevant models of the phases/sub-phases. For example friction model during main phase of the process can be changed from dry friction to fluid friction. The CLF based model of the synchronizer can also be connected with rest of components models of the gearbox. For example the CLF based model can also be linked with gear transmission models or the bearing models. Before solving the equations of motion by using the numerical algorithm, the sleeve is considered as a master and the gear is considered as a slave. Results are obtained with the sleeve and the ring axial and rotational movements, and the gear rotational movement. Validation of modified form of the generic synchronizer with available experimental test rig predicted reasonable accuracy of the developed mathematical model of using CLF. Effect of sleeve vibrational motion, cone angle, cone coefficient of friction and shift force on synchronization time is also analyzed. Different studies and analysis can be performed through CLF based model. For example sensitivity analysis of the gear shifting mechanism, the mathematical model can be modified with new design of the gear shifting mechanism, can explain the upshift and downshift, can also be used for multi-objective optimization and can explain any range of speed of the gear shifting either from gear two to gear four or from gear five to gear six. In short a CLF based mathematical model is developed to explain in detail the complex motion of the gear shifting mechanism.

Paper B: Dynamics and Pareto Optimization of a Generic Synchronizer Mechanism

Further for the process of development of smooth, quick and energy efficient mechanism, kinematics and dynamics of the generic synchronizer are studied by using CLF model. Pareto optimization is performed to optimize rate of shift force, cone angle and cone coefficient of friction by using the genetic algorithm in Matlab. The synchronization time and speed difference are chosen as objective functions to be minimized. Before the optimization a sensitivity analysis is performed with two objective functions and three parameters. The objective functions have opposite trends with respect to the parameters. One of the outcomes of optimization is the minimal admissible cone angle of the synchronizer.

Paper C: Performance Improvement of the Transmission Synchronizer via Sensitivity Analysis and Parametric Optimization

Generally in this paper the degrees of influence of the design parameters are identified by optimization. The optimization routine is applied at different driving conditions. Time duration of the gear shifting for heavy vehicles sometimes exceeds the normal time duration without quality. The phenomenon of abnormal gear shifting is expected to occur more frequently during particular operating conditions. Besides the nominal operating conditions effect of vibrational motion and road grade are also studied on the gear shifting mechanism (synchronizer). Before starting optimization the generic synchronizer is modified according to the available test rig and validated.

Results show that the model can predict gear shifting process reasonably well. Validation also proofs flexibility of the model. The model can be modify according to the new design and still the model is able to predict the gear shifting process.

Synchronization time and speed difference are selected as objective functions of the synchronizer. Eight parameters (cone angle, cone coefficient of friction, cone radius, rate of shift force, blocker angle, blocker coefficient of friction, gear moment of inertia and ring moment of inertia) are chosen for study. Sensitivity analysis of the parameters with respect to the objective functions is performed. Some of the parameters are directly proportional to one objective function and inversely proportional to other objective function.

Matlab routine of the multi-objective optimization is applied to find out the optimized values of the parameters at minimum values of the objective functions. Eight different cases are considered with master, slave and both slaves settings of the sleeve and the gear. The eight cases are categorized into nominal, road grade and vibrational motion. Optimization routine is applied for each case. From graphical pictures of optimization results such trends of the parameters are identified at which the synchronizer performs optimally. The parameters are divided into higher degree of influence and lower degree of influence parameters. The lower degree influence parameters have no influence upon optimal performance of the gear shifting mechanism. Instead of taking eight parameters together only parameters of higher degree of influence are sufficient to measure performance of the synchronizer.

Paper D: Failure modes and optimal performance of a generic synchronizer

Another model of the gear shifting mechanism is developed in GT-Suite software. By using the model failure modes are predicted. Optimization is performed in GT-Suite software just to minimize the synchronization time. Parametric sensitivity analysis of the gear shifting mechanism is used to identify the failure modes. System response characteristics are monitored by varying the structural design parameters and control design parameter. Limits of values of the control design and structural design parameters are identified at which the synchronizer fails to perform by using the developed GT-Suite model of a gear shifting mechanism. Shift force as the control design parameter and blocking angle, indexing angle, cone angle, cone coefficient of friction, cone radius and spring force are selected as the structural design parameters. Sleeve and ring axial displacements, speed difference and relative rotation of the blocking teeth are chosen as system response characteristics. It is concluded that variations of the shift force predict the failure modes in terms of clashing. Variations of the blocker angle and the indexing angle predict failure modes in terms of blockage of the sleeve. Variations of the cone angle, the cone coefficient of friction, the cone radius and the spring force predict failure modes in terms of the longer synchronization time.

Optimization routine of GT-Suite model is applied with the genetic algorithm to identify values limits of the control design and structural design parameters at which the synchronizer can perform optimally. Reasonable bounds of the parameters are implemented with initial values to run the optimization routine of the GT-Suite model. The control design and structural design parameters are considered as independent parameters and the synchronization time is considered as an objective function to be minimized. The control design and structural design parameters are plotted against the synchronization time to draw valuable conclusions. It is concluded generally that at lower values of the shift force, the cone angle, the cone radius and the cone coefficient of friction and at higher values of the blocker angle, the indexing angle and the spring force, the synchronizer perform optimally in sense of minimum synchronization time. Analyses of performance of the gear shifting mechanism at optimized values of the parameters shows that it could not be possible practically to achieve such a less synchronization time because the gear drastically gains rotational speed almost near to end of the process. To void such phenomenon number of optimizations are performed at different particular values of the shift force (1000 N, 1250 N, 1500 N, 1750 and 2000 N). It is found that optimized performance of the gear shifting mechanism can be obtained at lower value of the shift force by using optimized values of the structural design parameters.

Paper E: Minimizing synchronization time of a gear shifting mechanism by optimizing its structural design parameters

In addition to the studies in previous papers seventeen structural design parameters are considered for optimization in GT-Suite software to minimize the gear shifting time under different driving conditions. Two cases as sample cases are shown in detail. In first case sleeve is considered as a master at vibrational motion and in second case gear is considered as a master at road grade. Six cases of master/slave at nominal, road grade and vibrational motion are chosen for optimization. Optimization is performed for all six cases with initial guess of parameters as input. Synchronization time is selected as objective function and seventeen structural design parameters are chosen as input parameters. Optimized values of the structural design parameters are obtained for each case. Synchronization time is improved by 50 percent from the time at initial guess to the time at optimized values. At average optimized values in three cases the time is almost 23 percent less and in rest of three cases the time is 41 percent less. As a result the average values of optimized parameters can be considered as robust values.

10 Conclusions and outlook

10.1 Conclusions

The developed mathematical model based on CLF can predict the gear shifting process. The model is validated by experimental setup (paper A and C). The developed model can not only describe in detail the synchronization process by sub phases but can also provide the opportunity to implement other relevant mathematical models. For example the friction model used in [4] is applied in the mathematical model. The synchronization time (quickness) and the speed difference (smoothness) are considered as objective functions in the sensitivity analysis. Parameters: cone angle, cone coefficient of friction, applied shift force, blocker angle, blocker coefficient of friction, cones radius, gear moment of inertia and ring moment of inertia are considered as input parameters. It is predicted from the sensitivity analysis that both objective functions have same trends with increasing cone radius, gear moment of inertia and ring moment of inertia but both objectives have opposite trends with increasing rest of the parameters. Because of the conflicting behavior between quickness and smoothness, Pareto optimization is performed using a Matlab routine of multi-objective optimization with genetic algorithm. In addition to optimal values of the parameters it is also found from Pareto optimization analysis that the robust gear shifting process can be achieved by just taking into account the parameters which have higher degree of influence instead of taking all effecting parameters. The most influencing parameters are different in different cases with different settings of master/slave and with different conditions of vibrations, road grade and nominal. For instance applied shift force, cones angle, cones coefficient of friction, blocker angle and blocker coefficient of friction are the highest influencing parameters where the sleeve is considered as a master at nominal condition. At conditions of vibrations the ring moment of inertia replaces the blocker angle and at condition of road grade the cones radius replaces the cones angle among the highest influencing parameters. But some of the parameters are common in some cases which have highest degree of influence. For example the applied shift force is among the highest influencing parameters in the cases where the gear or the sleeve is considered as a master. In short the research work has contributed a mathematical model of the synchronization process and found the parameters with their optimal values which can give optimal performance of the gear shifting process.

GT-Suite software is used to identify failure modes and optimized values of the system parameters. A model of the gear shifting mechanism is developed in GT-Suite software. Failure modes are identified via sensitivity analysis. Four system response characteristics are plotted against the time and used to identify the failure modes which are sleeve and ring axial displacements, speed difference and relative rotation of the blocking teeth. Limits of the parameters values are identified where the mechanism fails to perform. Optimization routine of the GT-Suite is applied on the model. At first hand eight parameters are taken into account for optimization. Later on after the first optimization results it is concluded that the optimized performance can also be obtained without taking shift force as input parameter. Later on seven parameters are taken into account as independent variables and synchronization time as an objective function. Percentage changes of the variables from their initial values are calculated and analyzed. Finding of average optimal values of parameters of the gear shifting mechanism is valuable contribution to design reliable and efficient transmission system for automotive industry especially for heavy vehicles. Because average optimized values of the parameters can also minimize the gear shifting time at different driving scenarios instead of considering different optimized values for different driving scenario.

Structural design parameters must be known to manufacture a gear shifting mechanism. Therefore at final stage of the project almost all available structural design parameters of the mechanism in GT-Suite model are taken for study. Optimization is performed with same settings and algorithm as used before. The synchronization time is considered as an objective function and seventeen parameters are considered for optimization. These parameters are engaging teeth left and right angle, blocking teeth left and right angle, indexing angle, cone angle, cone coefficient of friction, blocking teeth friction, engaging teeth friction, minimum and maximum ring radius, blocking teeth radius, engaging teeth radius, ring mass, gear mass, ring moment of inertia and gear/sleeve moment of inertia. Six cases at different settings of master/slave with nominal, road grade and vibrational motion are considered. Optimal values of the parameters are obtained in every case. The optimal synchronization time, 0.022 sec, in all cases is almost 50 percent less than the initial synchronization time, 0.044±0.004 sec. In practical applications there must be such a robust synchronizer which can perform at all conditions. So, average optimized values of the structural design parameters from all six cases are considered for the robust synchronizer. The synchronizer performance diagrams are obtained for six cases at average optimized parameters values. The synchronization time at average values for these six cases is higher than the synchronization time in each case but still much less than the initial time. In two cases the time is 0.034 sec, in one case the time is 0.029 sec and in three cases the time is 0.026 sec at average values. Optimal values of the parameters are obtained from optimization at worst conditions of road grade and excitation with 50 Hz frequency and 134 rpm amplitude. The performance diagrams are obtained at lowering speed of 195 rpm of road grade effect, at 38 rpm amplitude and 100 Hz frequency. The synchronization time is still almost same as at worst conditions. So it is concluded that the average optimized values of the structural design parameters are robust values.

The study has provided answers of the questions which were asked in the section 1.1 (research focus and questions).

- The flexible mathematical model is developed based on CLF to explain the complex gear shifting process (paper A and B).
- Degrees of influence of the structural design parameters upon the gear shifting process are identified (paper C).
- By the GT-Suite developed model of the gear shifting mechanism, the limits of the parameters values where the mechanism fails to perform are identified (paper D).
- Within working range of values of the parameters, their average optimized values are identified for different driving conditions (paper E).

10.2 Future work

In addition to modelling the gear shifting mechanism, other transmission systems components (gearbox, bearings etc) could be modelled by using constrained Lagrangian formalism. Effects of the failure modes of gear shifting upon different bodies (frictional cones, engaging teeth) could be studied. Focusing upon the gear shifting mechanism of electric vehicles could also be a good idea by using the developed models.

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Part II

Appended Papers A-E